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AERODYNAMIC INVESTIGATION OF FLOW THROUGH A CENTRIFUGAL COMPRESSOR STAGE

R.Rajendran*[1], S.Ramamurthy*[2] and P.Mohanan*[3]
*[1] Scientist, Propulsion Division, N.A.L, Bangalore–560 037, India. raghavanrajendran@gmail.com
*[2] Scientist, Propulsion Division, N.A.L, Bangalore–560 037, India. ramamurthy_srm@yahoo.com
*[3] Professor, Dept. of Mech. Engg.,NITK, Mangalore-575 025, India. profmohanan@hotmail.com

ABSTRACT: *The overall efficiency of the compressor is dependent on the design of both the impeller and diffuser. The vane diffuser reduces the operating rangas compared to vaneless diffuser. However, by proper setting of the diffuser with reference to impeller, it is possible to achieve a good stage performance. This paper describes the experimental investigation of the detailed flow behavior inside a centrifugal compressor stage for three different setting angles of the vane diffuser with reference to the fixed impeller blade outlet angle. It is seen that diffuser setting angles lower than the impeller outlet flow angle gives higher stage efficiency as well as flow range.*

1. INTRODUCTION

Centrifugal compressor stage performance is dependent not only on the impeller aerodynamics, but also on the aerodynamics of the vaned diffuser following the impeller. The vaned diffuser influences the flow at impeller outlet. To extend the operating range of centrifugal compressor, it is necessary to develop a quantitative understanding of the basic flow phenomenon between impeller and diffuser. In a high pressure ratio centrifugal compressor a vane diffuser is provided after the impeller to convert large kinetic energy into useful static pressure head. This is accomplished through diffusion process. As the vane diffuser has both a semivaneless and vaned portions. The flow in the semivaneless portion experiences large variations in the velocity and pressure in radial and circumferential directions.

Usually the vaned diffuser will be set at an angle equal to impeller absolute outlet flow angle to cover the operating range. The leading edge is appropriately modified to handle different flow rates to meet the operating range. However in the actual practice, the flow to the diffuser will be approaching at a different flow angle than the impeller exit flow angle. The present work involved an understanding, how the diffuser setting affects the total characteristics of the stage.

Experimental investigations in a high-pressure ratio centrifugal compressor stage consisting of impeller and vaned diffuser with three different blade setting angles of 70, 65 and 60 degrees with reference to the radial direction were carried out by detailed steady and unsteady measurements using conventional and high response miniature pressure transducers. Time averaged and unsteady flow measurements were carried out at different speeds ranging from 15000 to 20000 rpm in a closed circuit centrifugal compressor test rig.

2. EXPERIMENTAL SET UP

Test Facility

The layout of the Closed Circuit Centrifugal Compressor Test Rig (CLOCTER) is shown in Fig.1. An electromechanically coupled twin DC motor system rotates the compressor at a desired speed. Thyristor control with feedback for the DC motors ensures maintenance of the speed to an accuracy of 1%. The compressor and DC motor were connected together with a step up gear box (1:20). An electronic torque meter coupled in between the gear box and the compressor was used to measure compressor speed and input power. A gate valve provided in the closed circuit was used to vary the mass flow rate through the compressor. An orifice plate in the inlet duct was used for mass flow measurement. A heat exchanger in the closed circuit was used to ensure steady inlet flow conditions.

Test Compressor

The assembly view of the compressor stage with vane diffuser configuration is shown in Fig.2, which consists of a backswept impeller of 300 mm tip diameter having 19 blades and a vane diffuser having 17 blades. The vane diffuser instrumented blades for the static pressure measurements at suction and pressure surfaces are shown in Fig.3.

Instrumentation

The time averaged parameters like total pressure, static pressure, total temperature, speed and power input to the compressor were measured using conventional probes and an on-line data acquisition system and electronic torquemeter.

The static pressure fluctuations at different flow coefficients were measured using high response miniature Kulite transducers at impeller shroud located at different positions in impeller channel and also at diffuser channel. The block diagram of the instrumentation system for unsteady pressure measurement using Kulite transducer is shown in Fig.4.

3. RESULTS AND CISCUSSIONS

The performance characteristics of the impeller and the compressor stage were obtained by running the compressor in closed circuit with air as working medium. Different operating points were obtained for speeds ranging from 15000 to 20000 rpm in steps of 1000 rpm by varying the mass flow rate using a main and an interconnecting gate valve. The pressure ratio and mass flow rates were estimated from the measured time averaged parameters. The performance characteristics of the impeller and stage with three different configurations of the diffuser are shown in Fig.5 and Fig.6. It is observed from this figure that the operating range varies with the diffuser blade setting angles. The pressure ratios are maximum at D1 reduces at D2 and minimum at D3. This phenomenon indicates that as the operating region increases the pressure ratio reduces at all speeds. The pressure recovery in vane diffuser at D1, D2 and D3 configurations are plotted in Fig.7 for different speeds. It is observed from this figure that the diffuser with D3 configuration has the higher pressure recovery coefficient and the pressure recovery coefficient more or less remains same for wider operating range. It is observed that, with decrease in setting angle the performance of the diffuser improves.

The impeller and stage efficiencies at all the three configurations of the diffuser are shown in Fig.8 for speeds. It is observed from this figure that, with decrease in setting angle the peak impeller total-to-total efficiency decreases, but at lower setting angle of the diffuser the stage efficiency is higher over wider mass flow rate and the peak stage efficiency do not change with setting angle.

The static pressure measurements in the diffuser channel are measured at different radius from diffuser leading edge to diffuser exit in all the three configurations and are plotted for four different mass flow rates at 20000 rpm and are shown in Fig.9. It is observed from this figure, for a given setting angle the static pressure inside the diffuser channel increases with decrease in mass flow rate. This increase in static pressure at a given mass flow rate is higher for lower setting angle. Since the included angle between the diffuser increases, this provides higher diffusion.

The compressor stage provides continuous increase in static pressure from inlet to outlet; these characteristics will depend on the operating point and the speed of the compressor. To study this, the static pressures were measured on the shroud from impeller inlet to diffuser exit. The static pressure variation at different location on the shroud and diffuser channel is shown in Fig.10. In this figure, the static pressure variations from impeller leading edge to diffuser exit are shown for 20000 rpm for all the three configurations of the diffuser. It is observed at largest mass flow the static pressure peaks in the vaneless portion and then rises constantly till diffuser trailing edge, at low mass flow rate the static pressure drops in the vane less portion and increases constantly in D1 and D2 configurations, whereas in configuration D3 at large flow rate the static pressure drops in the vaneless portion and increases constantly till diffuser exit indicating flow acceleration. In D2 configuration there is a small increase in static pressure at the impeller inlet and the magnitude increases at low flow rates. This behavior is observed in all the speeds. This is a common phenomenon observed in radial flow machines that at low flow rates there is a small separation region at impeller inlet tip.

The vaned diffuser provides static pressure recovery in a compressor stage. This recovery depends on the incidence to the diffuser. To study this, the static pressures were measured at suction and pressure surface wall of the vaned diffuser in three configurations of the diffuser at different speeds and mass flow rates. Variations of the static pressure along the blade length for four mass flow rates at 20000 rpm are plotted in Fig.11. At high flow rates the incidence to the diffuser is positive and the flow tends to separate from the suction surface, due to this we see there is a small cross over in the pressure distribution and as the flow rate reduces this cross over vanishes and the diffuser provides the required loading which gives rise to increase in static pressure. It is also seen very close to the trailing edge both the pressure s becomes equal. Similar observations were made at other setting angles

The flow inside the centrifugal impeller is highly unsteady and this unsteady flow is being carried out into the diffuser and affects the performance. To study this, static pressure measured on the shroud wall was made using high response transducers. Four different transducers were mounted on the impeller shroud to cover one blade channel from inlet to outlet. Similarly another four different transducers were mounted in diffuser shroud wall to cover the diffuser channel. Fig.12 shows the static pressure fluctuation at impeller and diffuser at D2 configuration for 20000 rpm. In Fig.12, the transducer signals for different mass flow parameter were shown along with locations. By the side of each figure, the numerical value which gives the normalized mean static pressure at that location. It is observed from the figure, that at large mass flow the flow at the leading edge of the impeller is highly unsteady and it gets attenuated when it reaches the impeller exit. The attenuated unsteady gets amplified up to the mid channel of the diffuser and becomes uniform at the diffuser exit. At low mass flow close to the design the unsteady flow generated in the leading edge of the impeller remains more or less same magnitude at the exit and gets amplified in the diffuser leading edge and gets attenuated as it goes towards the exit. The average static pressure increases from impeller inlet to the diffuser exit.

4. CONCLUSIONS

Diffuser setting angle influences the impeller performance and with lower setting angle the impeller and stage efficiency decreases. Diffuser setting angles lower than the impeller outlet flow angle gives higher stage efficiency as well as range. With decrease in setting angle the peak impeller total-to-total efficiency decreases but at lower setting angle, the stage efficiency is higher over wider mass flow and the peak stage efficiency do not change with setting angle. The vaned diffuser faces a highly non uniform flow with amplitudes being higher at low mass flow rates and as the flow reaches the exit it becomes uniform.

REFERENCES

[1] Fink.D.A "Unsteady flow phenomenon in Centrifugal Turbomachinery", Final Technical Report, Gas Turbine Lab. Publication, Report No.AD-A179714, 1986. 4P.

[2] Kubo.T, Murata.S, Unsteady flow phenomenon in centrifugal fans" JSME International Journal, Vol19, No.135, September 1996, P 1039-1046.

[3] Ramamurthy.S, Murugesan.K, "Unsteady flow phenomenon in centrifugal impeller", VI National convention of Aerospace Engineers, 1991, P 16.

Fig.1 CLOCTER Test Facility

Fig.2 Compressor Stage

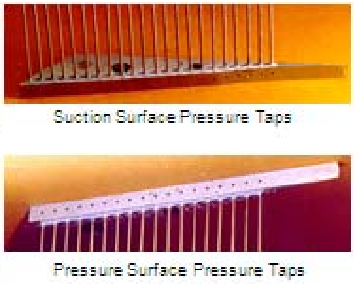

Fig.3 Vane Diff. Instrumented Blades

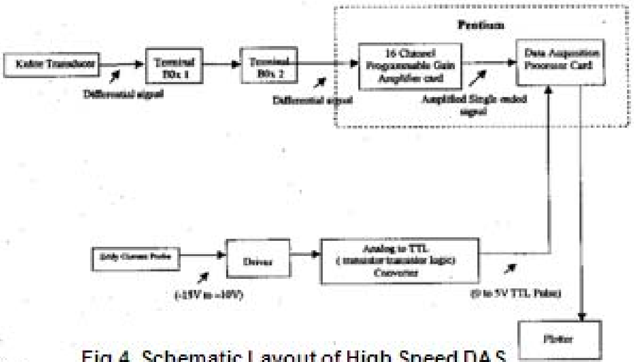

Fig.4 Schematic Layout of High Speed DAS

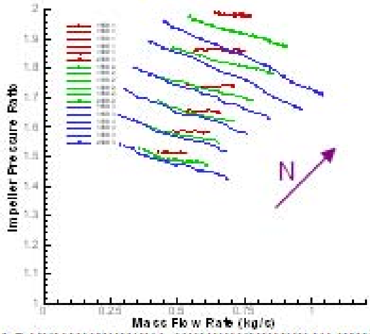

Fig.5 Performance Characteristics of Impeller

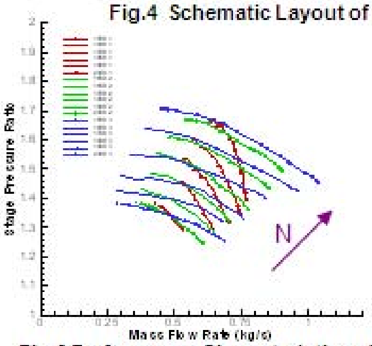

Fig.6 Performance Characteristics of Stage

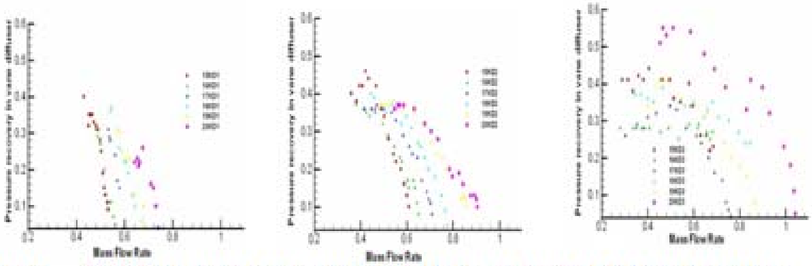

Fig.7 Pressure Recovery in Vane Diffusers at D1, D2 & D3 Configurations

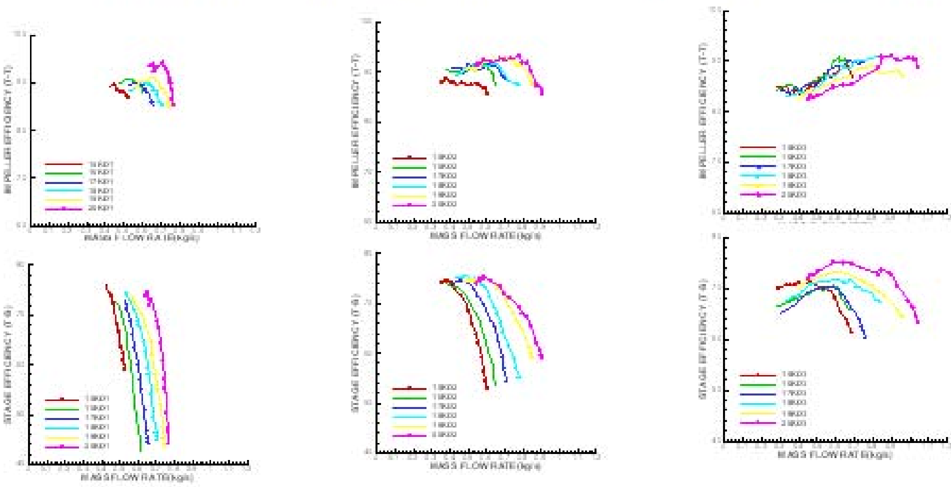

Fig.8 Impeller and Stage Efficiencies at D2 Configuration

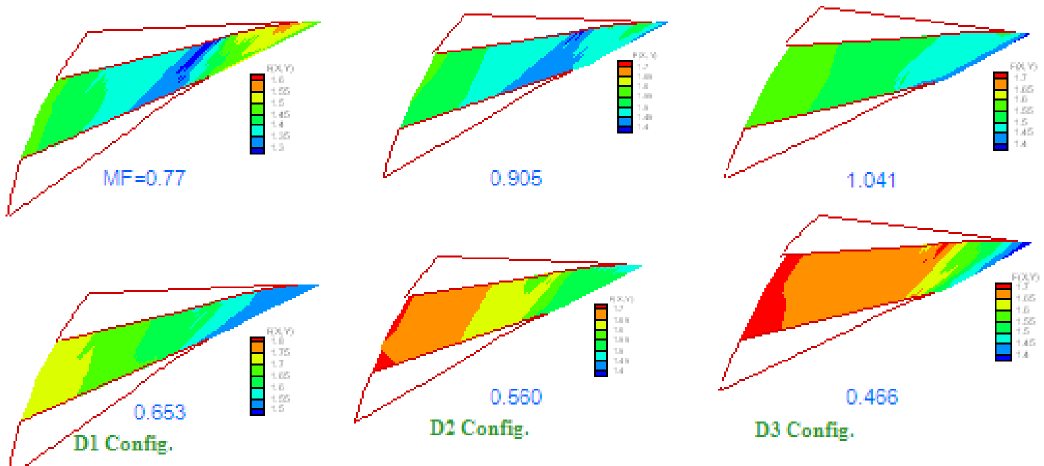

Fig.9 Static Pressure Distribution in Diffuser Channel at 20000 rpm

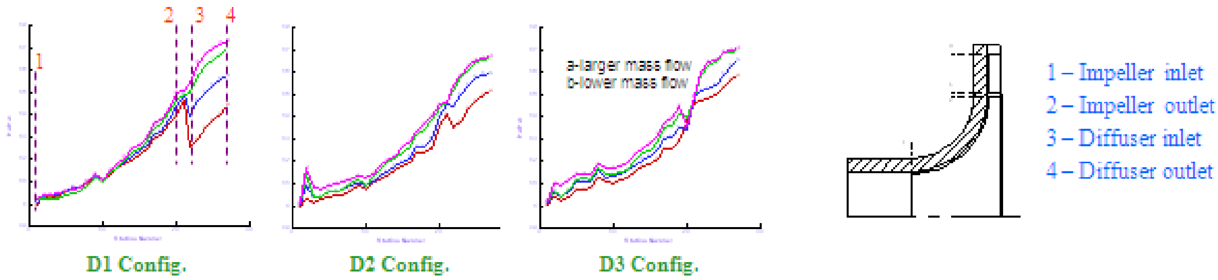

Fig.10 Static Pressure Variation from Impeller Inlet to Diffuser Exit at 20000 rpm

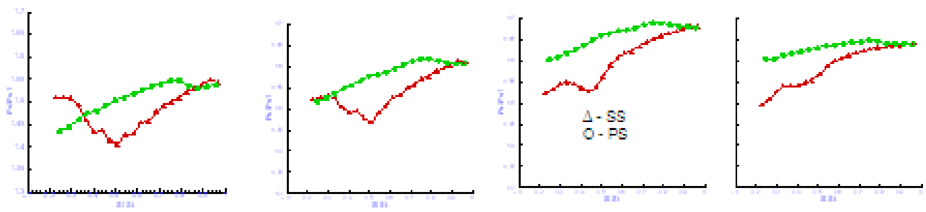

Fig.11 Static Pressure Variation at Suction and Pressure Surface of Diffuser D2 at 20000 rpm

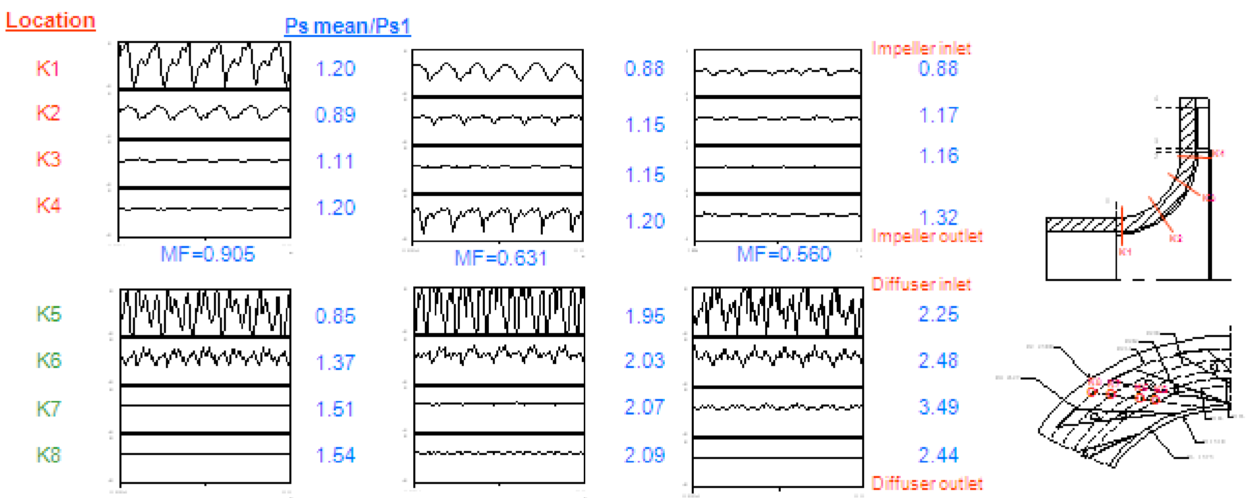

Fig.12 Static Pressure Fluctuations at Impeller and Vaned Diffuser D2 at 20000 rpm